Notice No.7

Rules and Regulations for the Classification of Ships, July 2020

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Please note that corrigenda amends to paragraphs, Tables and Figures are not shown in their entirety.

Issue date: June 2021

Amendments to	**Effective date**	**IACS/IMO implementation (if applicable)**
Part 3, Chapter 9, Section 6	Corrigendum	N/A
Part 3, Chapter 13, Section 2	Corrigendum	N/A
Part 3, Chapter 14, Section 1	Corrigenda	N/A
Part 4, Chapter 8, Section 2	Corrigenda	N/A
Part 5, Chapter 2, Section 3	Corrigenda	N/A
Part 5, Chapter 16, Section 2	Corrigendum	N/A
Part 5, Chapter 10, Section 2	Corrigendum	N/A
Part 5, Chapter 11, Section 1	Corrigendum	N/A
Part 6, Chapter 1, Section 2	Corrigendum	N/A
Pt 7, Chapter 9, Section 5	Corrigendum	N/A

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Part 3, Chapter 9
Special Features

■ *Section 6*
Lifting appliances and support arrangements

6.6 Walk-to-Work (W2W)

6.6.2 The transfer equipment is to be certified by LR against appropriate code relevant for the equipment, i.e. LR's ~~*Code for Lifting Appliances in a Marine Environment*~~ *Code for Offshore Personnel Transfer Systems*~~, as applicable or an alternative code deemed applicable by LR~~.

Part 3, Chapter 13
Ship Control Systems

■ *Section 2*
Rudders

2.12 Connections of rudder blade structure with solid parts

2.12.3 The minimum section modulus, Z_s, of the cross-section of the structure of the rudder blade formed by vertical web plates and rudder plating, which is connected with the solid part where the rudder stock is housed is to be not less than:

~~$Z_s = c_s {d_c}^3 \left(\frac{H_E - H_X}{H_E}\right)\left(\frac{k}{k_s}\right) \times 10^{-4} \text{ cm}^3$~~ $Z_s = c_s {d_c}^3 \left(\frac{H_E - H_X}{H_E}\right)^2 \left(\frac{k}{k_s}\right) \times 10^{-4} \text{ cm}^3$

2.15 Cone couplings with special arrangements for mounting and dismounting the couplings

2.15.4 The push-up pressure, P_{req} is not to be less than the greater of the two following values:

It has to be proved by the designer that the push-up pressure does not exceed the permissible surface pressure in the cone. The permissible surface pressure is to be determined by the following formula:

~~$P_{perm} = \frac{0{,}95\,\sigma_g(1-\alpha^2)}{\sqrt{3+\alpha^4}} P_b \text{ N/mm}^2$~~ $P_{perm} = \frac{0{,}95\,\sigma_g(1-\alpha^2)}{\sqrt{3+\alpha^4}} - P_b \text{ N/mm}^2$

Part 3, Chapter 14
Cargo Securing Arrangements

■ *Section 1*
General

1.3 Application of the BoxMax (V, W, L) notation

1.3.5 The Master needs to select the appropriate significant wave height for the limited duration voyage, H_{SL}. The significant wave height for the limited duration voyage is to be rounded up to the next integer and is to be selected as follows:

H_{SL} = ~~$H_{foercast}$~~ $H_{forecast}$ if the T_{arrive} is less than 72 hours after the time of the initial forecast.

= $H_{forecast}$ + 2 m if the T_{arrive} is more than 96 hours after the time of the initial forecast.

Part 4, Chapter 8
Container Ships

■ *Section 2*
Materials

2.3 Requirements for the use of thick steel plates

Table 8.2.1 Preventative measures to be used in design and construction for thick steel plates

Note 2. Measure 2 may be required where enhanced NDE during construction has been applied as part of Measure ~~4~~ 3

2.3.13 Where enhanced NDE is used to achieve Measure 3 *see ~~Pt 4, Ch 8, 2.3 Requirements for use of thick steel plates 2.3.9~~ Pt 4, Ch 8, 2.3 Requirements for use of thick steel plates 2.3.10(d)*, the block to block butt welds of the hatch coaming side plate, hatch coaming top plate and upper deck are to have a minimum crack tip opening displacement (CTOD) value of 0,18 mm. The CTOD tests to confirm the CTOD value are to be carried out to the satisfaction of LR.

Part 5, Chapter 2
Reciprocating Internal Combustion Engines

■ *Section 3*
Crankshaft Design

3.2 Scope

3.2.9 Further information and guidance on crankshaft design is provided in the LR~~’s~~ *Guidance Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*.

3.3 Information to be submitted

3.3.2 The following information is also required for appraisal of the crankshaft (not contained in Form 2073):

- every surface treatment affecting fillets or oil holes shall be specified so as to enable calculation according to Chapter ~~2~~ 3 of the LR *Guidance Notes for ~~Crankshaft SCF Calculation using Finite Element Method~~ the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*;
    - o this is to include ~~Crankshaft~~ crankshaft fatigue enhancement factors ~~K_1~~ K_1 and ~~K_2~~ K_2 where applicable.

3.4 Symbols

3.4.1 For the purposes of this Chapter the following symbols apply, *see also*;

D_A = the outside diameter of we or twice the minimum distance between ~~centre line~~ centreline of journals and outer contour of web, whichever is less, in mm

W_{eqw} = ~~xection~~ section modulus related to cross-section of web, in mm^3

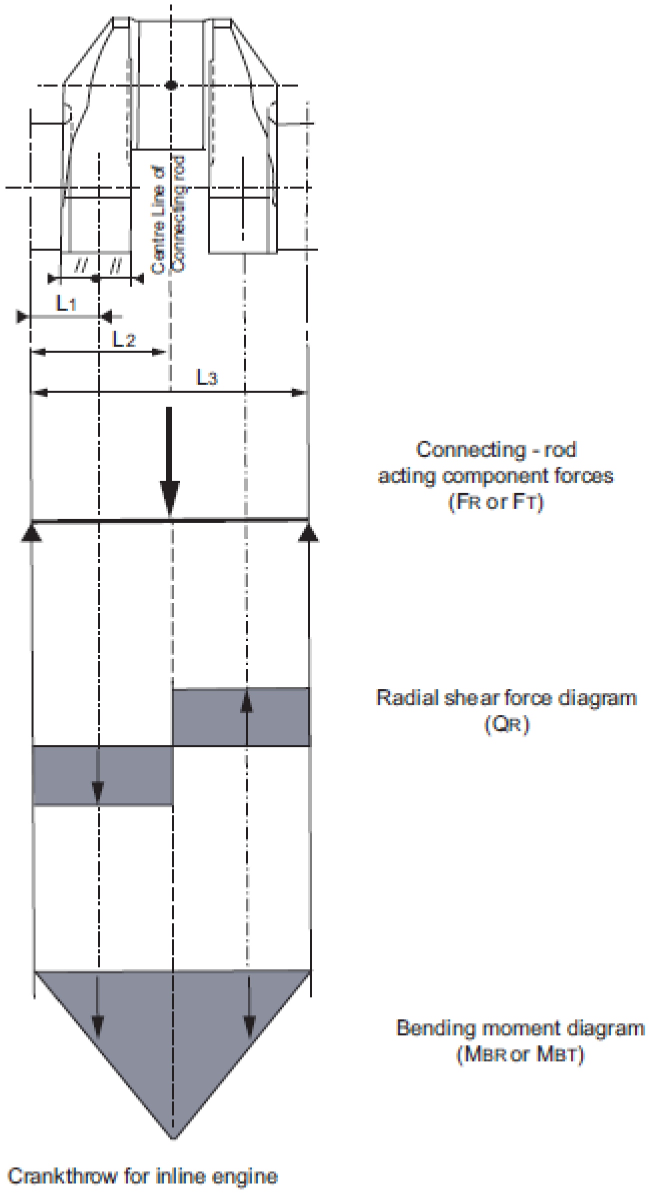
y = distance between the adjacent generating lines of journal and pin, in mm

Note for $y \geq 0{,}5D_s$ ~~where~~. Where y is less than $0{,}1D_s$, special consideration is to be given to the effect of the stress due to the shrink fit on the fatigue strength at the crankpin fillet.

σ_y = equivalent alternating stress for crankpin fillet, journal fillet or outlet of crankpin oil bore as applicable, in ~~N/mm^3~~ N/mm^2

3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions

Figure 2.3.9 Bending moment and shear force for in-line engine crankthrows has been replaced with the below figure;


Centre Line of Connecting rod
//
//
L1
L2
L3
Connecting - rod acting component forces (FR or FT)
Radial shear force diagram (QR)
Bending moment diagram (MBR or MBT)
Crankthrow for inline engine

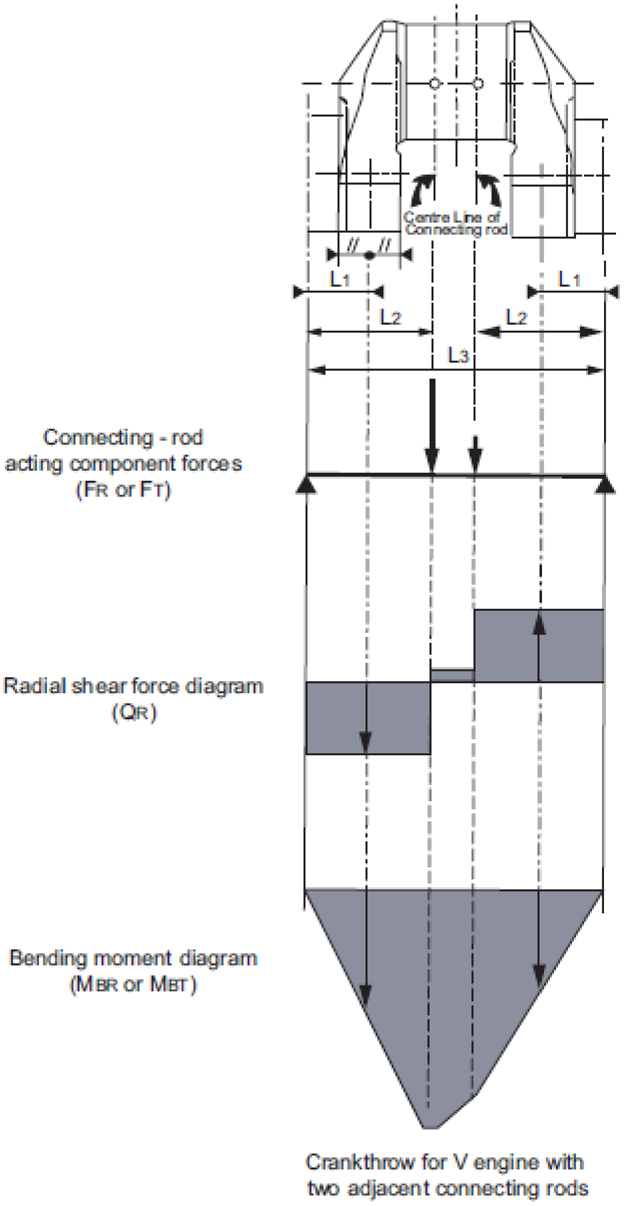
L_1 = Distance between main journal centreline and crankweb centre

L_2 = Distance between main journal centreline and connecting - rod centre

L_3 = Distance between two adjacent main journal centrelines

Figure xxxx Bending moment and shear force for in-line engine crankthrows

Figure 2.3.10 Bending moment and shear force for V engine crankthrows has been replaced with the below figure;


Centre Line of Connecting rod
L1
L1
L2
L2
L3
Connecting - rod acting component forces (FR or FT)
Radial shear force diagram (QR)
Bending moment diagram (MBR or MBT)
Crankthrow for V engine with two adjacent connecting rods

L_1 = Distance between main journal centreline and crankweb centre
L_2 = Distance between main journal centreline and connecting - rod centre
L_3 = Distance between two adjacent main journal centrelines

Figure 2.3.10 Bending moment and shear force for V engine crankthrows

3.5.4 ~~The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore. *See Figure 2.3.9 Bending moment and shear force for in-line engine crankthrows* and *Figure 2.3.10 Bending moment and shear force for V engine crankthrows*.~~ ~~M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected~~

The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore. *See;*

- *Figure 2.3.5 Crankpin section through the oil bore*,
- *Figure 2.3.9 Bending moment and shear force for in-line engine crankthrows*, and
- *Figure 2.3.10 Bending moment and shear force for V engine crankthrows.*

M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected.

3.6 Calculation of bending stresses

3.6.1 The radial and tangential forces due to gas and inertia loads acting upon the crankpin at each connecting-rod position are to be calculated over one working cycle. Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments, M_{BRF}, M_{BRO} and M_{BTO}, and radial forces, Q_{RF}, as defined in *Pt 5, Ch 2, Calculation of alternating stresses due to bending moments and radial forces – assumptions ~~3.5.2~~ 3.5.3* and *Pt 5, Ch 2, Calculation of alternating stresses due to bending moments and radial forces – assumptions ~~3.5.3~~ 3.5.4* are then calculated.

3.6.6 Nominal alternating bending and compressive stresses in a web cross-section are calculated as follows:

$$\sigma_{BFN} = \pm\frac{M_{BRFN}}{W_{eqw}}10^3 \; K_e \; \text{N/mm}^2$$

$$\sigma_{QFN} = \pm\frac{Q_{RFN}}{F} \; K_e \; \text{N/mm}^2$$

where

~~$M_{BRFN} = \pm\frac{1}{2}(X_{BRF\,max} - X_{BRF\,Min})\text{Nm}$~~ $M_{BRFN} = \pm\frac{1}{2}(M_{BRF\,max} - M_{BRF\,min})\text{Nm}$

~~$Q_{RFN} = \pm\frac{1}{2}(Q_{RF\,max} - Q_{RF\,Min})\text{Nm}$~~ $Q_{RFN} = \pm\frac{1}{2}(Q_{RF\,max} - Q_{RF\,min})\text{N}$

3.6.7 Nominal alternating bending stress in the outlet of the crankpin oil bore is calculated as follows:

$$\sigma_{BON} = \pm\frac{M_{BON}}{W_e}10^3 \; \text{N/mm}^2$$

where

M_{BON} is taken as the ½ range value $M_{BON} = \pm ½ (M_{BOmax} – M_{BOmin})$ Nm

3.7 Calculation of torsional stresses

3.7.1 The nominal alternating torsional stress, ~~τ_n~~ τ_N, is to be taken into consideration. The value is to be derived from forced-damped vibration calculations of the complete dynamic system. Alternative methods will be given consideration. The engine designer is to advise the maximum level of alternating vibratory stress that is permitted (τ_a).

3.7.2 τ_a or ~~τ_n~~ τ_N(as applicable) is to be applied as a limiting value for the torsional vibration assessment required by *Pt 5, Ch 8, 2 Torsional vibration*.

3.7.3 Nominal alternating torsional stress is calculated as follows:

~~τ_n~~ $\tau_N = \frac{M_{TN}}{W_P}10^3$ N/mm^2

where

~~$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right)$ mm^3 for the crankpin, or $W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_c}\right)$ mm^3 for the jorunal~~

$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right)$ mm^3 for the crankpin, or $W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_c}\right)$ mm^3 for the journal

3.8 Stress concentration factors

3.8.3 Where the geometry of the crankshaft is outside the boundaries (*see Table 2.3.2 Crankshaft variable boundaries for analytical SCF calculation*) of the analytical SCF the calculation method detailed in Chapter 1 and Chapter 4 of the LR *Guidance Notes for ~~Crankshaft SCG Calculation using Finite Element Method~~ Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts* may be undertaken.

3.8.5 Chapters1 and ~~3~~ 4 of the LR *Guidance Notes for ~~Crankshaft SCG Calculation using Finite Element Method~~ Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts* describe how finite element (FE) analyses can be used for the calculation of the SCF. Care needs to be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

3.8.6 Crankpin SCF are calculated as follows:

a) Bending

~~$\alpha_B = 2{,}6914 f(s,w).f(w).f(b).f(r).f(d_G).f(d_H).f(\text{recess})$~~

$\alpha_B = 2{,}6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(\text{recess})$

b) Torsion

~~$\alpha_T = 0{,}8 f(r,s).f(b).f(w)$~~

$\alpha_T = 0{,}8 \cdot f(r,s) \cdot f(b) \cdot f(w)$

where

$f(b) = 7{,}8955 - 10{,}654b + 5{,}3482b^2 – 0{,}857b^3$~~$f(w) = w^{(-0{,}145)}$~~

$f(w) = w^{(-0,145)}$

3.8.7 Journal fillet SCF are calculated as follows (not applicable to semi-built) crankshafts):

a) Bending

~~$\beta_B = 2{,}7146 f_B(s,w).f_B(w).f_B(b).f_B(r).f_B(d_G).f_B(d_H).f(\text{recess})$~~

$\beta_B = 2{,}7146 \cdot f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(\text{recess})$

b) Compression due to the radial force~~:~~

~~$\beta_Q = 3{,}0128 f_Q(s).f_Q(w).f_Q(b).f_Q(r).f_Q(d_H).f(\text{recess})$~~

$\beta_Q = 3{,}0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(\text{recess})$

3.9 Additional bending stress

3.9.1 In addition to the alternating bending stresses in fillets (*see Pt 5, Ch 2, 3.6 Calculation of bending stresses 3.6.8*) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying ~~σ_{add}~~ σ_{add} as given by *Table 2.3.3 Additional bending stresses*.

3.10 Equivalent alternating stress

3.10.4 Equivalent alternating stress, σ_V, is defined as:

(a) For the crankpin fillet:

~~$\sigma_V = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3\tau_G{}^2}\ \text{N/mm}^2$~~ $\sigma_V = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3\tau_H{}^2}\ \text{N/mm}^2$

(b) For the journal fillet:

~~$\sigma_V = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3\tau_H{}^2}\ \text{N/mm}^3$~~ $\sigma_V = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3\tau_G{}^2}\ \text{N/m}$

3.11 Fatigue strength

3.11.1 The fatigue strength is to be understood as the value of equivalent alternating stress (~~Von~~ von Mises) which a crankshaft can permanently withstand at the most highly stressed points. The fatigue strength can be evaluated by means of the following formulae:

a) Related to the crankpin diameter:

~~$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0,2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_E}\sqrt{\frac{1}{R_X}}\right]\ \text{N/mm}^2$~~

$$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0,2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right]\ \text{N/mm}^2$$

b) Related to the journal diameter:

~~$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D_G{}^{-0,2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_E}\sqrt{\frac{1}{R_G}}\right]\ \text{N/mm}^2$~~

$$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0,2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right]\ \text{N/mm}^2$$

A value for K_2 will be assigned upon application by the engine designers. Full details of the process, together with the results of full scale fatigue tests will be required to be submitted for consideration. *See* Chapter 2 of the LR *Guidance ~~note - Guidance for the evaluation of Crankshaft Fatigue Tests~~ Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*. Alternatively, the following values may be taken (surface hardened zone to include fillet radii):

3.11.3 Fatigue strength calculations or, alternatively, fatigue test results determined by experiment based either on full size crankthrow (or crankshaft), or on specimens taken from a full size crankthrow, may be required to demonstrate acceptability. The experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment are to be submitted for approval by LR. The procedure is to include as a minimum: method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, and confidence number. *See also* Chapter 2 of the LR *Guidance ~~for the Evaluation of Crankshaft Fatigue Tests~~ Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts.*

3.11.5 Only surface treatment processes approved by LR are permitted. Guidance for calculation of surface treated fillets and oil bore outlets is presented in Chapter ~~2~~ 3 of the LR *Guidance Notes for ~~Crankshaft SCF Calculation using Finite Element Method~~ the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*.

3.13 Shrink fit of semi-built crankshafts

3.13.4 The maximum permissible internal diameter in the journal pin is to be calculated in accordance with the following formula, this condition serves to avoid plasticity in the hole of the journal pin:

$$D_{BG} = D_S \sqrt{1 - \frac{4000 S_R M_{max}}{\mu \pi D_S^2 L_S \sigma_{SP}}} \text{mm}$$

where

M_{MAX} = absolute maximum value of the torque M_{Tmax} in accordance with ~~*Pt 5, Ch 2, 3.13 Shrink fit of semi built crankshafts 3.13.7*~~ *Pt 5, Ch 2, 3.7 Calculation of torsional stresses 3.7.3*, in Nm.

3.13.5 The actual oversize Z of the shrink fit must be within the limits Z_{min} and Z_{max} calculated in accordance *Pt 5, Ch 2, 3.13 Shrink fit of semi-built crankshafts* ~~*3.13.7*~~ *3.13.6* and *Pt 5, Ch 2, 3.13 Shrink fit of semi-built crankshafts* ~~*3.13.8*~~ *3.13.7*. When *Pt 5, Ch 2, 3.13 Shrink fit of semi-built crankshafts 3.13.4* cannot be complied with, then the calculated values of Z_{min} and Z_{max} are not applicable due to multizone-plasticity problems. In such cases Z_{min} and Z_{max} are to be established from FEM calculations.

3.13.6 The minimum require diametral ~~Interference~~ interference is to be taken as the greater of:

$$Z_{min} \geq \frac{\sigma_{SW} D_S}{E_m} \quad \text{mm}$$

and

$$Z_{min} \geq \frac{4000}{\mu\pi} \frac{S_R M_{max}}{E_m D_S L_S} \frac{1 - Q_A^2 Q_S^2}{\left(1 - Q_A^2\right)\left(1 - Q_S^2\right)} \quad \text{mm}$$

where

Q_A = ~~shaft~~ web ratio, $Q_A = \frac{D_S}{D_A}$

Q_S = ~~web~~ shaft ratio, $Q_S = \frac{D_{BG}}{D_S}$

Part 5, Chapter 11
Other Pressure Vessels

■ *Section 1*
General requirements

1.2 Definition of symbols

1.2.1 The symbols used in the various formulae in *Pt 5, Ch 11, 2 Cylindrical shells and drums subject to internal pressure* to *Pt 5, Ch 11, 7 Standpipes and branches* inclusive, unless otherwise stated, are defined as follows, and are applicable to the specific part of the pressure vessel under consideration:

c = corrosion allowance in mm for ~~sign~~ design and operating condition taking into account the intended life cycle; a minimum corrosion allowance of 0,75 mm is to be used, any deviation is to be agreed with LR taking into consideration the material type, service fluid, design and operating conditions. The corrosion allowance is to be added to the calculation Rule thickness as well as the minimum thickness specified by the Rules.

Part 5, Chapter 16
Water Jet Systems

■ *Section 2*
General requirements

2.1 Water jet arrangement

2.1.1 In general, for a ship to be assigned an unrestricted service notation, a minimum of two water jet systems are to be provided where these form the sole means of propulsion. For ships where a single water jet system is the sole means of propulsion or steering, a detailed engineering and safety justification is to be evaluated by LR, *see Pt 5, Ch 16, 2.3 Calculations and information ~~2.3.22~~ 2.3.23*. This evaluation process will include a risk assessment analysis using a recognised technique to verify that sufficient levels of redundancy and monitoring are incorporated in the water jet unit's essential support systems and operating equipment.

2.1.2 ~~w~~Water jet propulsion units are to be capable of continuous operation between their maximum and minimum output power rating at specified operating conditions, *see Pt 5, Ch 1, 3 Operating conditions* and within the operational service profiles defined by *Pt 5, Ch 16, 2.3 Calculations and information 2.3.11* and *Pt 5, Ch 16, 2.3 Calculations and information 2.3.12*.

Part 5, Chapter 19
Steering Systems

■ *Section 2*
Performance

2.1 General

2.1.1 Main and auxiliary steering gear for rudder-type steering systems are to meet the performance requirements in *Pt 5, Ch 19, 2.2 Performance requirements for rudder-type steering systems*. Steering performance on ships fitted with steerable podded propulsion systems is to comply with *~~Pt 5, Ch 16, 3.7 Nozzle/steering arrangements~~ Pt 5, Ch 9, 5.6 Steering system* in addition to the requirements of this Chapter, as applicable. Steering performance on ships fitted with steerable water jet systems is to comply with *Pt 5, Ch 16, 3.7 Nozzle/steering arrangements* in addition to the requirements of this Chapter, as applicable. Steering performance on ships fitted with azimuth thruster systems is to comply with *Pt 5, Ch 20, 2.1 General* in addition to the requirements of this Chapter, as applicable

Part 6, Chapter 1
Control Engineering Systems

■ *Section 2*
Essential features for control, alarm, monitoring and safety systems

2.10 Programmable electronic systems – General requirements

2.10.20 Software lifecycle activities, e.g. design, development, supply and maintenance, are to be carried out in accordance with an acceptable quality management system which has lifecycle models suitable to the nature of the software project, considering its size, complexity, safety, risk and integrity. Project specific software quality plans are to be submitted. These are to demonstrate that the provisions of ISO/IEC 90003: Software engineering – *Guidelines for the application of ISO 9001:~~2008~~ 2015 to computer software*, or equivalent, are incorporated. The plans are to define responsibilities for the lifecycle activities, including verification, validation, software module testing and, integration with other components or systems and security policies to be applied.

■ *Section 5*
Integrated Bridge Navigation System – IBS notation

5.2 General requirements

5.2.2 Where *Pt 7, Ch 9, 5.2 General requirements 5.2.1(a)* is applicable, the submissions required by *Pt 7, Ch 9, 1.2 Information and plans required to be submitted 1.2.1* are to include evidence demonstrating satisfaction of the requirements of an identified relevant Standard.

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